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(54) Title: METHOD AND DEVICE FOR THE DAMPING OF VIBRATION IN A CUTTING TOOL
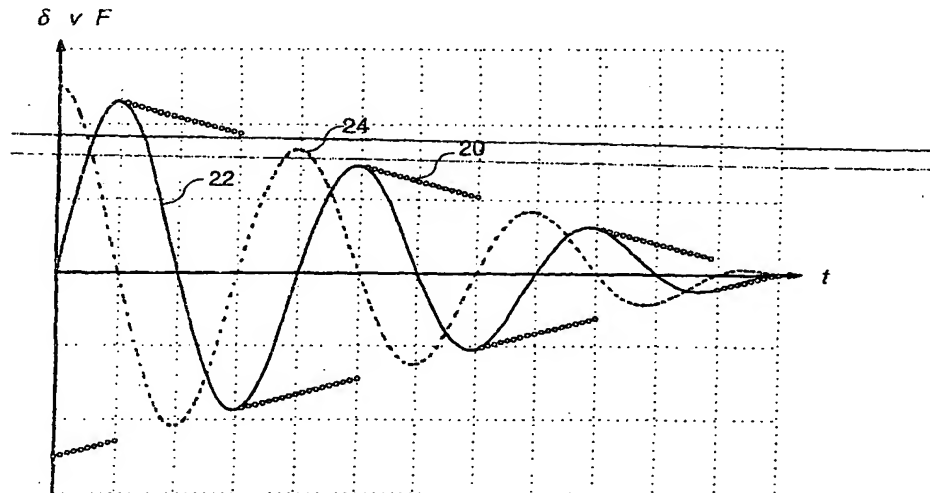
(57) Abstract: A method in order to control, by means of a control device, a damping force of a vibration damping device intended to be arranged in connection with a metallic tool for the chip removing machining of metal and to damp feed-back vibrations therein, including to detect the tool's oscillatory motion by means of sensor device associated with said control device, identify the oscillatory motion's phase by means of the control device, by means of the control device actuate said vibration damping device, so that the same renders a mechanical damping force with substantially the same frequency as the tool's oscillatory motion. According to the invention the control device is arranged to generate a damping force counter-directed the tool's velocity. The invention relates even to a metal tool for chip removing machining and a vibration damping device for the same.

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METHOD AND DEVICE FOR THE DAMPING OF VIBRATION IN A CUTTING TOOL

Technical Field of the Invention

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The present invention relates to a method to control, by means of a control device, a damping force of a vibration damping device intended to be arranged in connection with a metallic tool for the chip removing machining of metal and to damp feed-back vibrations therein, including

10   -   to detect the tool's oscillatory motion by means of sensor device associated with said control device,

      -    to identify the oscillatory motion's frequency, amplitude and phase by means of the control device,

      -    by means of the control device actuate said vibration damping device, so that the

15   same renders a mechanical damping force with substantially the same frequency as the mechanical structure's oscillatory motion.

It also relates to such a vibration damper and such a mechanical structure.

20   ## Background
Metallic tools for chip-forming machining can be exposed to vibrations induced by forces or regenerative oscillations (vibrations); see "Metal Cutting Theory and Practice" by Stephenson and Agapiou; publisher Marcel Dekker Inc.; ISBN: 0-8247-9579-2. According to this document vibrations induced by forces are generated by transient

25   cutting forces, whilst regenerative vibrations occur because the dynamic cutting process forms a closed loop. The present patent application relates only to regenerative oscillations (even called self-induced or feedback oscillations or vibrations).

Damping of vibration in tools for chip removing machining has previously been

30   achieved by pure mechanical damping, the shaft being formed with a cavity in which a counter-oscillating mass of for instance heavy metal is applied. In doing so, the weight and location of the mass is tuned in order to provide damping of oscillations within a

certain range of frequencies. The cavity is then filled with a viscous liquid, e.g. oil, and is plugged. However, this technique works passably only in those cases where the overhang of the shaft from a fastening device is approx. 4-10 times longer than the diameter thereof. In addition to this limitation, the pure mechanical damping has an

5 obvious disadvantage so far that the range of frequencies, within which the damping acts, is very limited. An additional inconvenience consists of the strength-wise weakening that the cavity formed in the shaft entails.

In entirely other areas of technology, the development of more efficient, adaptive

10 damping techniques based on the utilization of, among other things, piezo elements has been started. A piezo element consists of a material, most often of a ceramic type, which on compression or elongation in a certain direction - the direction of polarization - generates an electric field in the same direction. The piezo element usually has the shape of a rectangular plate having a direction of polarization, which is parallel to the

15 major axis of the plate. By connecting the piezo element to an electric circuit, including a control module, and compressing or elongating the piezo element in the direction of polarization, an electric current will be generated and flow in the circuit, electric resistive components included in the control module generating heat according to known physics. In doing so, vibration energy is converted to thermal energy, whereby a

20 passively damping, but not damping effect on the vibrations is obtained. Moreover, by forming the control module with a suitable combination of resistive and reactive components, so-called shunts, selected frequencies can be brought to be damped with particular efficiency. Advantageously, such frequencies are the so-called "own-frequencies" of the exposed "own-modes" of the object, which are those preferably

25 being excited.

Conversely, a piezo element may be compressed or elongated by an electric voltage being applied over the piezo element, during which the same may be used as a control or operating device (actuator). In that connection, this can be used for active vibration

30 control by selecting the polarity of the applied electric voltage in such a way that the mechanical stress of the operating device acts in the opposite direction, as an external, mechanical stress, the emergence of vibrations being suppressed by the fact that another

kinetic energy, for instance rotation energy, is prevented from being converted to vibration energy. In doing so, the synchronization of the applied electric voltage in respect of the external mechanical stress, the effect of which should be counteracted, takes place by a feedback signal from a deformation sensitive sensor being supplied a

5   control means in the form of a logical control circuit, e.g. a programmable micro-processor, in which the signal is processed to control the electric voltage applied over the operating device. The control function, i.e. the relation between the input signal from the sensor and the output voltage, may, in that connection, be made very complex. For instance, a self-learning system for adaptation to varying conditions is feasible. The

10   sensor may consist of a separate, deformation sensitive device, e.g. a second piezo element, or be common with the operating device.

Examples of practical applications and present development areas for the utilization of piezo elements for vibration damping purposes, are described in Mechanical

15   Engineering, Nov. 1995, p. 76-81. Thus, skis for alpine skiing (K2 Four ski, K2 Corp., USA) have been equipped with piezo·elements for the purpose of suppressing undesired vibrations, which otherwise decrease the contact with the ground, and thereby reduce the skier's prospects of a stable and controlled skiing. Furthermore, applications such as increased wing stability of aeroplanes, improved comfort in motor vehicles, suppression

20   of vibrations in helicopters' rotor blades and shafts, vibration control of process platforms for flexible manufacture, and increased accuracy of military weapons are mentioned. In information documents from Active Control eXperts (ACX) Inc., USA (manufacturer of piezo elements) vibration control of snowboards is also mentioned.

25   A method of the kind described in the introduction, as well as such a vibration damper and such a mechanical structure, respectively, is known from SE-A-9900441-8.

This type of vibration damper is not suitable for force induced vibrations, but only for regenerative, i.e. feed-back vibrations, which, e.g., arise in a tool during mechanical

30   machining when a small disturbance gives a mechanical feed-back in the tool. Such a mechanical feed-back may cause an increasing oscillatory motion, and thereby an undesired uneven surface of the machined blank and reduced service life of the tool.

In SE-A-9900441-8, it is not explicitly mentioned how the damping force should be applied on the mechanical structure. The hitherto known way to quench an oscillatory motion has, however, been to give a counter force in phase with the oscillatory motion.

5   This procedure works well as long as low frequencies are concerned. At higher frequencies, i.e. from approx. 500 Hz, it is difficult to apply a counter force without phase errors. If a phase error arises, there is a risk of the oscillatory motion and the damping force ending up in unbalance, and thereby partly amplifying each other, which in turn may lead to the oscillatory motion not being quenched to the desired degree.

10   Thus, a presumption for such a vibration damping to work is that the counter-directed force to a large degree of accuracy is in phase with the oscillatory motion.

Other piezoelectric dampers are described in SE-A-9803605-6, SE-A-9803606-4, SE-A-9803607-2, US-A-4 849 668, DE-A-199 25 193, EP-A-0 196 502,

15   US-A-5 485 053 and JP-A-63180401.

During chip removing machining, such as turning or drilling, it is not unrare for problems with vibrations to arise, particularly in cases in which the length of the shaft or tool outside the fastening device – a so-called overhang - is at least 3 times larger than the

20   diameter thereof. One type of vibration is bending vibration, the shaft being curved to and fro and submitted to bending deformations. This phenomenon constitutes a common problem, for instance during turning, especially internal turning, where the shaft in the form of a boring bar has to be long in order to reach the area in the workpiece which is to be machined, at the same time as the diameter of the bar is

25   limited by the dimension of the bore in which machining is carried out. During such drilling, turning and milling operations, where the distance to the workpiece is large, extenders are used, which frequently causes bending vibrations which not only lead to impaired dimensional accuracy and irregularities in the workpiece, but also to reduced service life of the tool and the cutting inserts or machining elements thereof.

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Summary of the Invention

The purpose of the present invention is to improve the control of a vibration-damping device for cutting tools having a regenerative oscillatory motion.

This has been attained by a method and a vibration damper, respectively, according to the kind described in the introduction, the control device being arranged to generate a damping force directed opposite to the cutting tool's velocity. This means that the damping force gives the tool a deformation directed in the opposite direction to that of the tool's motional velocity.

In this way, a damping of the regenerative motion is obtained without risk that the oscillations end up in imbalance, and thereby feed in more energy in the oscillatory motion, without any great demand for accuracy, as regards the phase displacement of the imposed damping force, i.e. the damping force should constantly resist the velocity and in order to obtain a maximum damping effect, a maximum damping force is imposed all the time. However, it is of minor importance with a maximum force at the oscillatory motion's end points, since the velocity there is low. Thus, it is important that the damping effect is greater than the contribution from the cutting process so that the regenerative oscillation is damped out and a smooth cutting process is obtained by means of the mechanical structure, e.g. a turning shaft or a boring bar.

Preferably, the control device is arranged to impose the damping force out of phase by 60°-120° alternatively 240°-300° in relation to the oscillatory motion. Suitably, the control device is arranged to impose the damping force out of phase by 70°-110° alternatively 250°-290° in relation to the oscillatory motion. Preferably, the control device is arranged to impose the damping force out of phase by 80°-100° alternatively 260°-280° in relation to the oscillatory motion. In this way, a faster damping of the oscillatory motion is obtained. Best results are achieved when the control device is arranged to impose the damping force out of phase by 90° alternatively 270° in relation to the oscillatory motion.

Preferably, the control device is arranged to actuate said vibration damping device so that the same renders a force, which is counter-directed to the velocity of said oscillatory motion. A co-directed force should be given when the same is imposed out of phase between 60° and 120° in relation to the oscillatory motion, while a counter-

5   directed force should be imposed out of phase between 240° and 300° in relation to the oscillatory motion.

Preferably, the control device is arranged to give a damping force in the area of 50-1500 Hz.

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Suitably, at least one piezo electric element is included in the vibration-damping device. Alternatively, the vibration-damping device may be a hydraulic or pneumatic cylinder or an electromagnetic device.

15   Preferably, said mechanical structure comprises a tool for chip removing machining.

Summary of Figures

In the following, the invention will be closer described, reference being made to the

20   accompanying drawings, where:

Figure 1 is a schematic side view of a long narrow body in the form of a tool shaft during bending deformation at oscillation (1st resonance frequency).

25   Figure 2 is a graph showing the bending torque in the body.

Figure 3 a side view of a cut end portion of the body in connection with a fastening end so as to illustrate the stress in the body during bending deformation proportional to elongation.

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Figure 4 is a transparent perspective view of a tool shaft.

Figure 5 is a perspective view of a bar extender for milling tools formed with a circular cross-section.

Figures 6-8 are perspective views of tool shafts having a square cross-section and in different alternative embodiments.

Figure 9 is a perspective view of a tool for active vibration damping mounted in a carrier.

Figure 10 is an analogous perspective view of an alternative embodiment for passive vibration damping.

Figure 11 shows schematically the damping of an oscillatory motion by means of a counter force in phase with the oscillation.

Figures 12-13 show schematically the damping of an oscillatory motion according to the invention.

Detailed Description

In figure 1, a long narrow body in the form of a bar or a shaft 1 of a tool is illustrated, which is intended to clamp one or more inserts during turning or milling. The body 1 has a fastening end 2 and a free, external end 3. The body has an external surface 4, which may consist of an envelope surface if the body is cylindrical. The same may also include a plurality of plane surfaces if the body has a polygonal, e.g. square cross-section shape. The body 1 may have an arbitrary cross-section shape, however, most commonly circular or square. In fig 1, numeral 5 designates a part in which the body 1 is fastened the body extending cantilever-like from the fastening part. In fig 1, the body 1 is shown in a state when the same has been deformed in a first bending "own-mode".

Furthermore, in figure 2 a graph is shown which illustrates how the bending torque M_b in this case varies along the body. As is seen in the graph, a maximum bending moment

arises, and thus a maximum elongation, at or near the fastening end 2. The same is valid for all lower modes, which are normally energy-wise dominant during bending vibrations of tools for chip removing machining.

5   In figure 3, a portion of the body 1 deformed by bending in fig 1 is shown in the area of the fastening end. In this connection, how the elongation at bending deformation varies in the cross-direction of the body (the elongation is strongly exaggerated for illustrative reasons) is illustrated. As is seen in the figure, the maximum elongations are obtained at the envelope surface or external surface 4 of the body.

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       In figure 4, a fundamental design of a bar or a shaft 1 is schematically shown at which two plate-formed, rectangular piezo elements 8 are fastened on opposite, longitudinal plane surfaces 4 of the shaft formed with square cross-section. The piezo elements 8 are placed in the area near the fastening end 2 of the shaft. At the external end 3 thereof, the

15   shaft has a machining element in the form of a cutting insert 9. Thus, the piezo elements 8 are positioned in an area where the maximum elongation occurs during bending deformation. Although this location is preferred, also other locations are feasible. Furthermore, the piezo elements 8 are oriented with the major faces thereof essentially parallel to the plane surfaces 4 of the bar or shaft 1 and with the major axes essentially

20   parallel to the length extension of the shaft or bar 1, at which the piezo elements 8 at bending vibration will be deformed while retaining the rectangular shape.

       In figure 5, an embodiment is shown according to which the body 1 consists of a bar extender intended for milling tool with a circular cross-section. In this case, a chip

25   forming machining element 9 in the form of a cutting edge is formed adjacent to a chip pocket 10 at the free end 3 of the bar extender. A piezo element 8 is attached on the envelope surface 4 of the bar extender in an area near the fastening end 2. The major axis of the piezo element is parallel to the length extension of the bar extender. Consequently, with this orientation, the piezo element 9 acts also here most efficiently

30   for the damping of bending vibration.

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For simultaneous damping of bending and torsional vibrations, the shaft of the tool is advantageously formed with a plurality of piezo elements, some of which being oriented with the long sides thereof essentially parallel to the length extension of the shaft, while others are oriented at approx. a 45 ° angle. Alternatively, one or more piezo element

5   have other orientations between said orientations.

Piezo elements are usually fragile, particularly such of the ceramic type. Therefore, in demanding environments, the same should have some form of protection in order to achieve an acceptable service life.

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In figures 6-8, tool shafts having a square cross-section are shown the piezo element 8 being attached and protected in alternative ways. In all cases, the piezo elements are placed in an area near the fastening part 5 (the same may consist of a conventional clamping unit in which the tool is detachably mounted).

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In Figure 6, the piezo element 8 is mounted in a countersink 11 and advantageously covered by a protection layer, for instance of the epoxy type.

In Figure 7, the piezo element is assumed to be mounted in the countersink 11 and

20   covered by a stiff lid 12.

In Figure 8, the piezo element 8 is mounted, e.g. glued on, on the outside of the shaft. These alternatives should only be seen as examples, those of which shown in figs 6 and 7 being preferred. It should be understood that the same type of protection for the piezo

25   elements is independent of the cross-section shape of the tool shaft.

The piezo elements co-operate with means for electric control or guiding of the same. In figures 9 and 10, examples are shown of how the tool 1 has been formed with such control means. In these cases, the tool is mounted in a carrier 13. In figure 10, a control

30   means for damping is shown in the form of a control module 14 formed near the fastening end 2 and an electric connection 15, via which one or more piezo elements 8 are connected to the control module 14 for separate or common control of respective

piezo element. This module 14 comprises at least electric resistive components.
Preferably, the control module 14 also comprises one or more shunts, at which selected
frequencies may be damped particularly efficiently.

5   Figure 9 illustrates a control means for active damping in the form of a free-standing
logical control circuit 16, e.g. a programmable micro-processor, for separate or common
control of (via the schematically illustrated electric connection 15) voltages applied over
the piezo elements 8. In practice, the connection 15 may in this case comprise collector
shoes or the like.

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Also if the piezo elements 8 in the embodiment exemplified in figure 10 for active
damping simultaneously act as both operating devices and sensors, it is feasible to
realise the same two functions by separate operating devices and sensors, at which the
sensors do not need to consist of piezo elements. Although the exemplified location of

15   the control module 14 and logic control circuit 16, respectively, is preferred, also other
locations are feasible. For instance, it is feasible, like the logic control circuit 16, to
form the control module 14 freestanding from the tool. The advantage of placing the
control module 14 in the vicinity of the fastening end is that the module becomes simple
to connect to the piezo elements, while the same at freestanding position becomes easier

20   to protect against harmful mechanical effects.

Through the usage of piezo elements as vibration dampers, a robust tool for chip
removing machining is obtained having a possibility of active damping of bending
vibrations over a wide range of frequencies. Furthermore, a tool is provided which on

25   one hand has a longer service life for the tool in itself as well as the cutting or
machining elements thereof, and on the other hand provides increased quality of the
surface on the machined workpiece. In addition, an improved working environment is
attained by the reduction of high frequency noise compared with previously known
tools.

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Figure 11 shows schematically how damping of an undesired oscillatory motion in a
mechanical structure generally comes about. A force 20 being counter-directed to and in

phase with the oscillatory motion 22 quickly quenches the same. However, this requires a very large accuracy as regards phase correctness. If a phase error arises, the counter-directed force will be partly co-directed to the motion, which may lead to the oscillatory motion not being quenched to the desired degree.

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Figure 12 shows schematically the damping of the oscillatory motion of the mechanical structure according to the invention.

The sensor detects the oscillatory motion. The signal is transferred to the control device,
10   which processes the signal and determines the oscillatory motion's phase by defining positive and negative, respectively, zero crossing. The control device also calculates the amplitude and frequency of the oscillation.

The control device then sends out a control signal to the vibration damper (actuator),
15   which generates a force counter-directed the tool's velocity. The phase is displaced a quarter of, alternatively three-quarters of, a wavelength in relation to the oscillatory motion.

In figure 13, it is shown how a decreasing counter force is imposed at decreasing
20   oscillation amplitude to avoid a new generation of vibrations, and is thereby easier to control.

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Claims

1. Method for controlling, by means of a control device, a damping force of a vibration damping device intended to be arranged in connection with a metallic tool for the chip removing machining of metal and to damp feed-back vibrations therein, including
   - to detect the tool's oscillatory motion by means of sensor device associated with said control device,
   - to identify the oscillatory motion's phase by means of the control device
   - by means of the control device actuate said vibration damping device, so that the same renders a mechanical damping force with substantially the same frequency as the tool's oscillatory motion, c h a r a c t e r i z e d in, that the control device is arranged to generate a damping force being counter-directed the tool's velocity.

2. Method according to claim 1, wherein the control device is arranged to impose the damping force out of phase by 60°-120°, alternatively 240°-300°, in relation to the oscillatory motion.

3. Method according to claim 1, wherein the control device is arranged to impose the damping force out of phase by 70°-110°, alternatively 250°-290°, in relation to the oscillatory motion.

4. Method according to claim 1, wherein the control device is arranged to impose the damping force out of phase by 80°-100°, alternatively 260°-280°, in relation to the oscillatory motion.

5. Method according to claim 1, wherein the control device is arranged to impose the damping force out of phase by 90°, alternatively 270°, in relation to the oscillatory motion.

6. Method according to any one of claims 1-5, wherein the control device is arranged to actuate said vibration damping device so that the same renders a force, which is counter-directed said velocity of the oscillatory motion.

7. Method according to any one of the preceding claims, wherein the control device is arranged to give a damping force in the range of 50-1 500 Hz.

5   8. Method according to any one of the proceeding claims, including that at least one piezoelectric element is included in the vibration-damping device.

9. Method according to any one of claims 1-7, including that at least one hydraulic or pneumatic cylinder is included in the vibration-damping device.

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10. Method according to any one of claims 1-7, including that at least one electromagnetic device is included in the vibration-damping device.

11. Vibration damper including a vibration damping device intended to be arranged in
15   connection to a metallic tool for chip removing machining of metals, a control device associated with said vibration damping device to control a damping force of said vibration damping device, and a sensor device associated with the control device arranged to detect the tool's oscillation, the control device being arranged to identify the amplitude, frequency and phase of the oscillation and to actuate said vibration damping
20   device, so that the same renders a mechanical damping oscillation with substantially the same frequency as the tool's oscillation, c h a r a c t e r i z e d in, that the control device is arranged to generate a damping force counter-directed the tool's velocity.

12. Vibration damper according to claim 11, wherein the control device is arranged to
25   impose the damping force out of phase by 60°-120°, alternatively 240°-300°, in relation to the oscillatory motion.

13. Vibration damper according to claim 11, wherein the control device is arranged to impose the damping force out of phase by 70°-110°, alternatively 250°-290°, in relation
30   to the oscillatory motion.

14.   Vibration damper according to claim 11, wherein the control device is arranged to impose the damping force out of phase by 80°-100°, alternatively 260°-280°, in relation to the oscillatory motion.

5   15.   Vibration damper according to claim 11, wherein the control device is arranged to impose the damping force out of phase by 90°, alternatively 270°, in relation to the oscillatory motion.

16.   Vibration damper according to any one of claims 11-15, wherein the control device
10   is arranged to actuate said vibration damping devices so that the same renders a force, which is counter-directed said velocity of the oscillatory motion.

17.   Vibration damper according to any one of claims 11-16, wherein the control device is arranged to give a damping force in the range of 50-1500 Hz.
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18.   Vibration damper according to any one of claims 11-17, wherein the vibration damping device comprises at least one piezo electric element.

19.   Vibration damper according to any one of claims 11-17, wherein the vibration
20   damping device comprises at least one hydraulic or pneumatic cylinder.

20.   Vibration damper according to any one of claims 11-17, wherein the vibration damping device comprises at least one electromagnetic device.

25   21. Tool of metal including a vibration damper according to any one of claims 11-20.
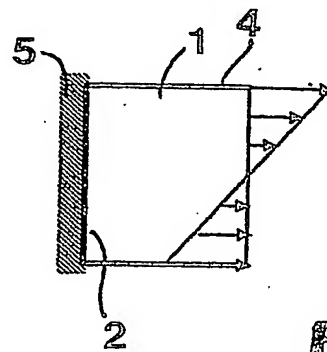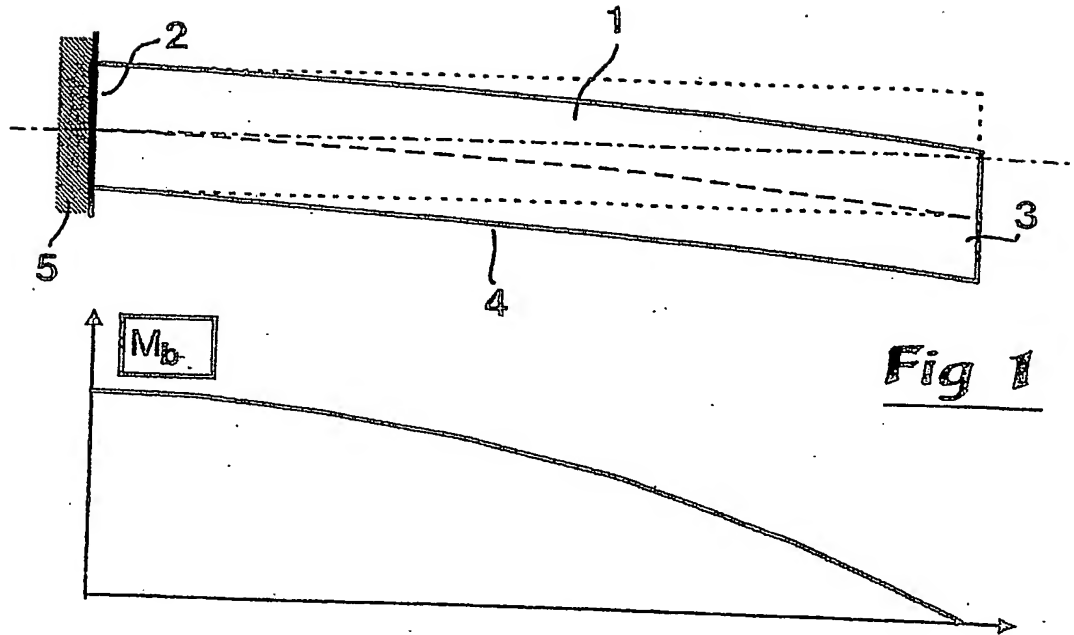
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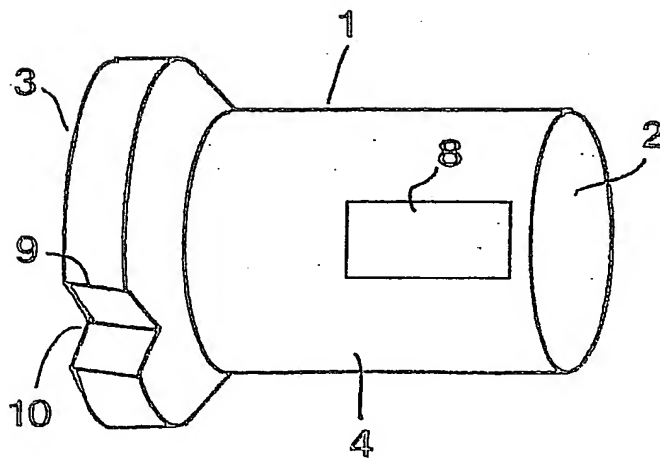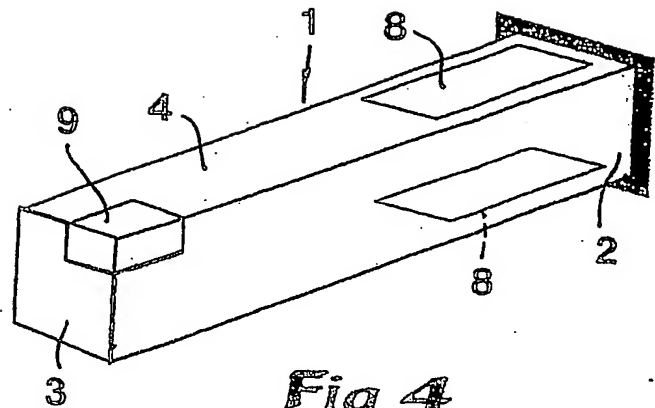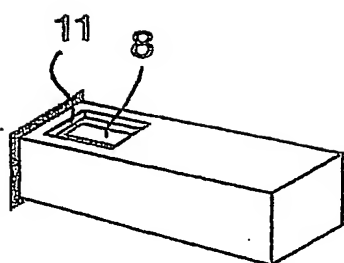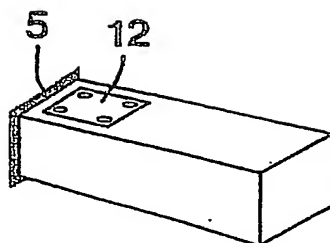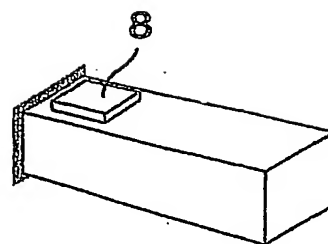
Fig 2

Fig 3

Fig 4

Fig 5

Fig 6

Fig 7

Fig 8

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Fig 9

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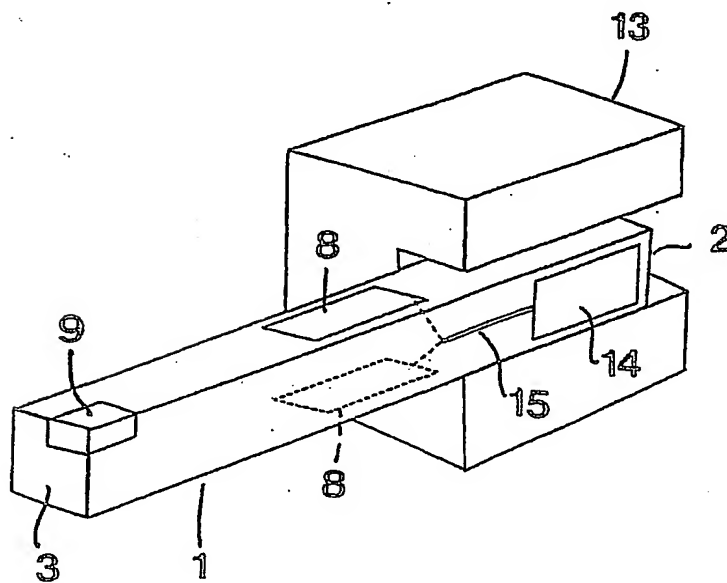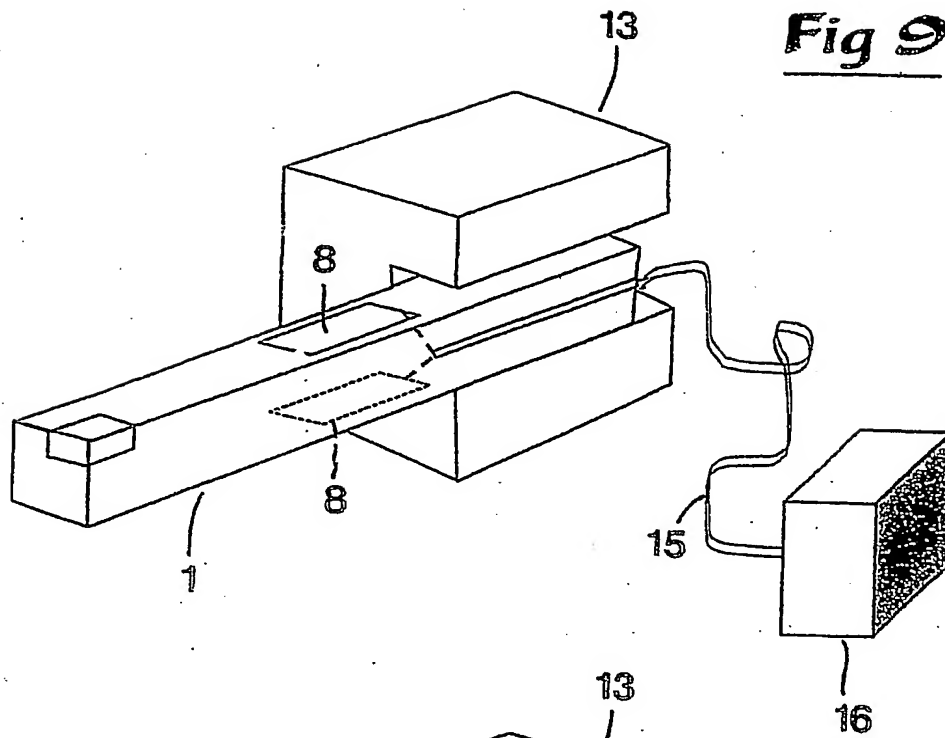
Fig 10

Fig 11
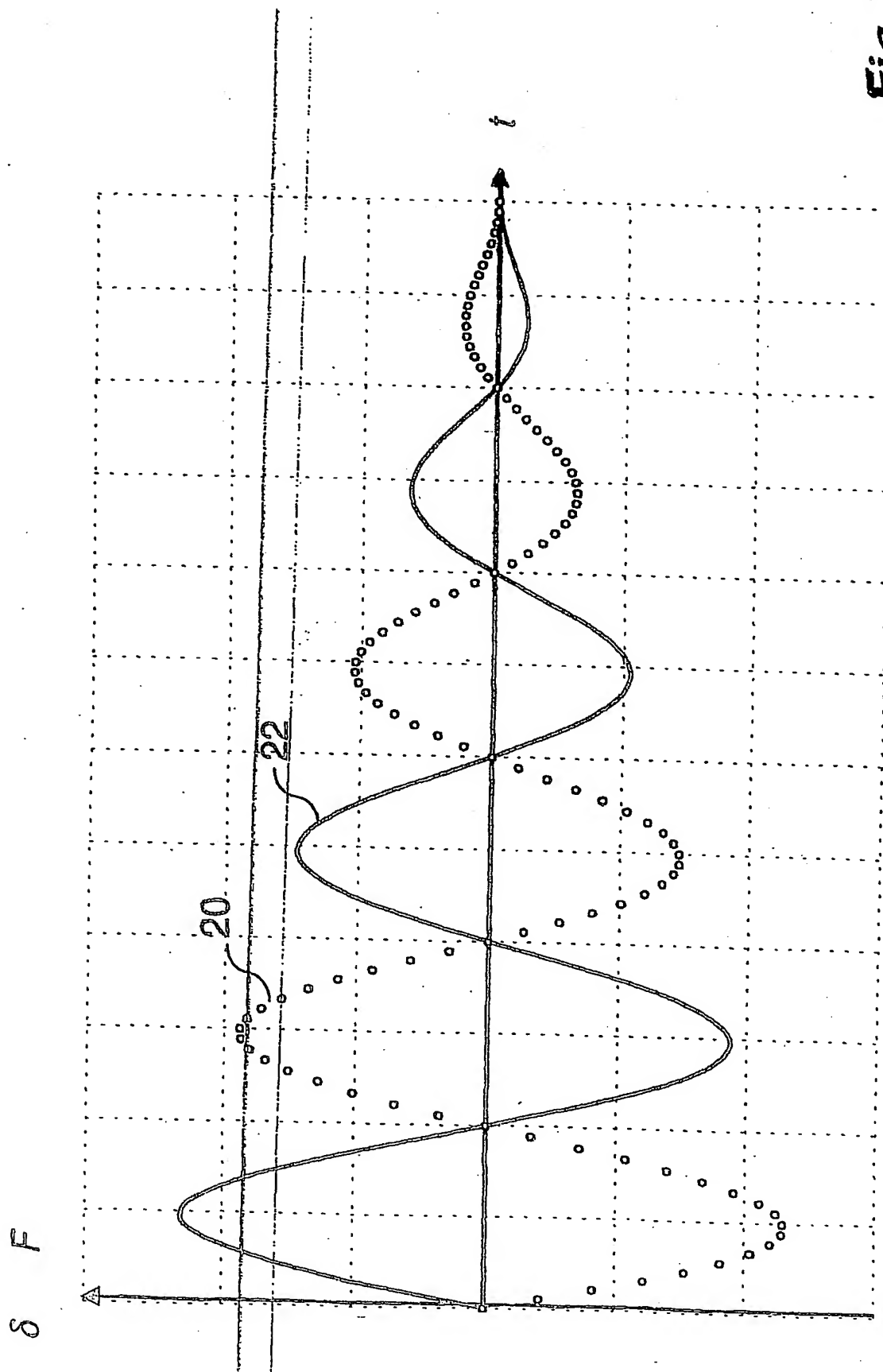
Fig 12
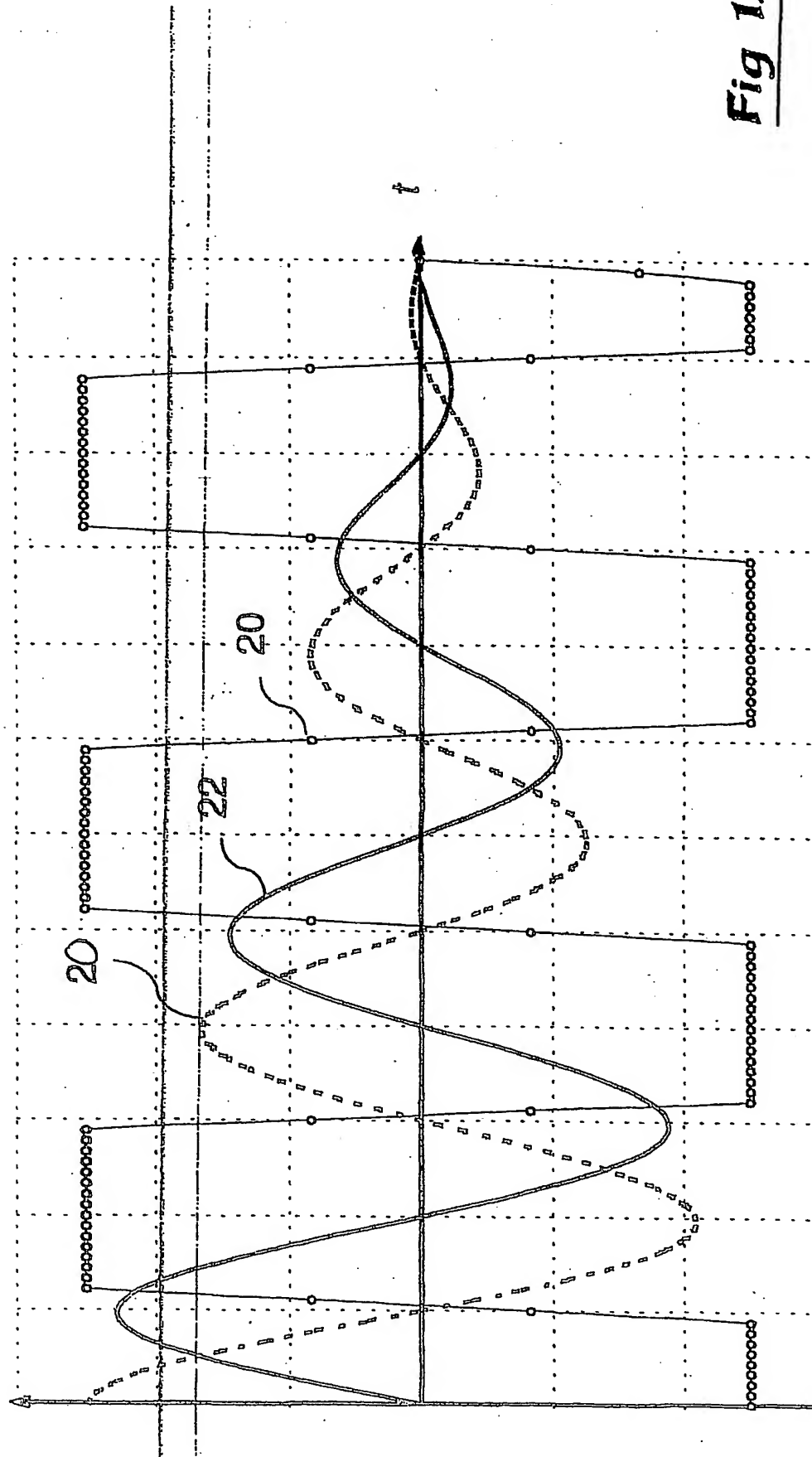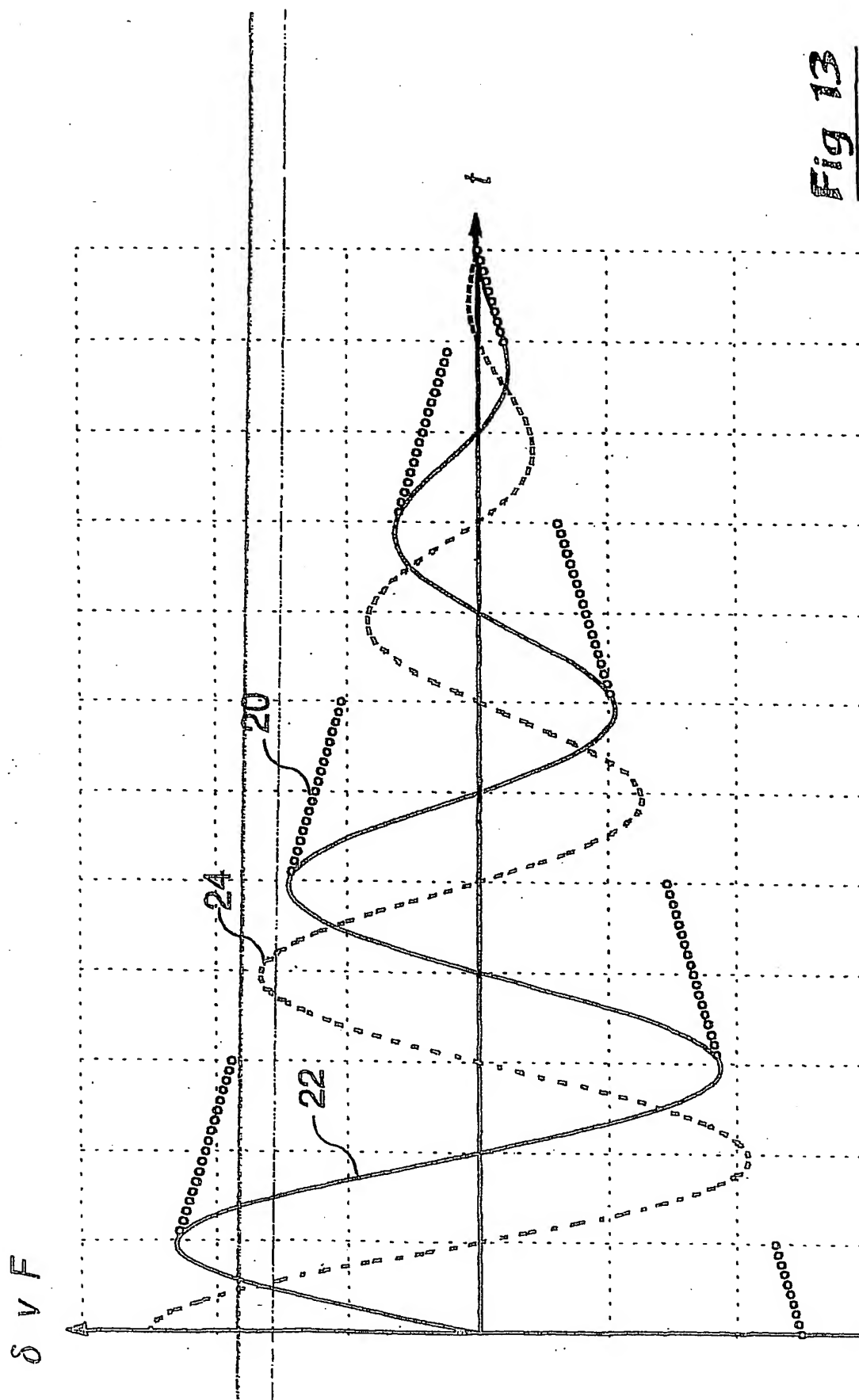
Fig 13

A. CLASSIFICATION OF SUBJECT MATTER

IPC7: B23B 29/12, B23Q 11/00

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC7: F16F, B23B, B23Q

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

SE,DK,FI,NO classes as above

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPO-INTERNAL, WPI DATA, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 5239789 A (N. UNO ET AL), 31 August 1993 (31.08.93) --	
A	EP 0297608 A1 (NISSAN MOTOR CO., LTD.), 4 January 1989 (04.01.89) --	
A	DE 2605476 A1 (KLEINWÄCHTER, J.), 18 August 1977 (18.08.77) -- --------	

☐ Further documents are listed in the continuation of Box C. ☒ See patent family annex.

Date of the actual completion of the international search	Date of mailing of the international search report
12 March 2002	18 -03- 2002
Name and mailing address of the ISA/ Swedish Patent Office Box 5055, S-102 42 STOCKHOLM Facsimile No. + 46 8 666 02 86	Authorized officer Annette Riedel / MRo Telephone No. + 46 8 782 25 00

Form PCT/ISA/210 (second sheet) (July 1998)

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US	5239789	A	31/08/93	CA	1331043	A	26/07/94
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				FR	2637663	A,B	13/04/90
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				JP	2102945	A	16/04/90
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				KR	9615417	B	13/11/96
				US	5182887	A	02/02/93
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EP	0297608	A1	04/01/89	DE	3867926	A	05/03/92
				JP	1009021	A	12/01/89
				JP	2568560	B	08/01/97
				US	4913409	A	03/04/90
DE	2605476	A1	18/08/77	NONE			